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(54) **Plate fin tube assembly and heat exchanger assembly employing same**

(57) A plate fin tube assembly consists of a plurality of plate fin tubes each of which comprises a plurality of spaced-apart plate members connected for heat conductive transfer to a tube. Each plate (60) is formed with a plurality of surface projections (86*a*, 86*b*) extending therefrom, which constitute fluid flow interference or vortex generator means. The projections may be formed in the plate by embossing or stamping, or may be formed on the plate by affixing strips thereto, Figs 4 to 6 (not shown).

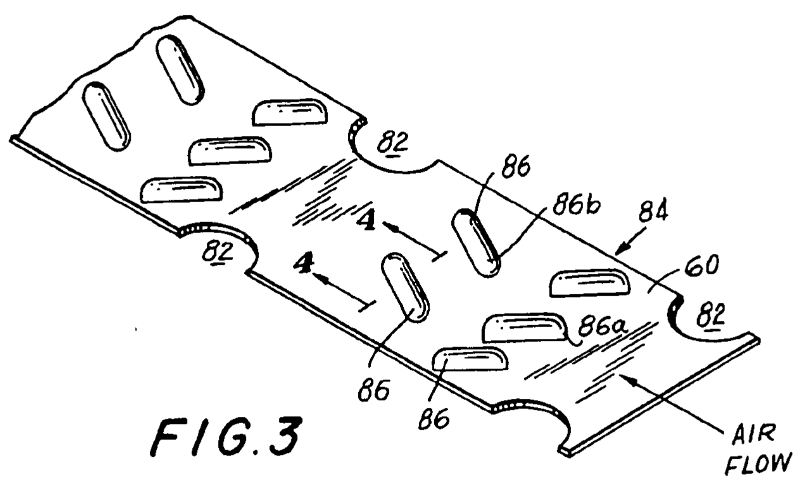

FIG.3

The drawings originally filed were informal and the print here reproduced is taken from a later filed formal copy.

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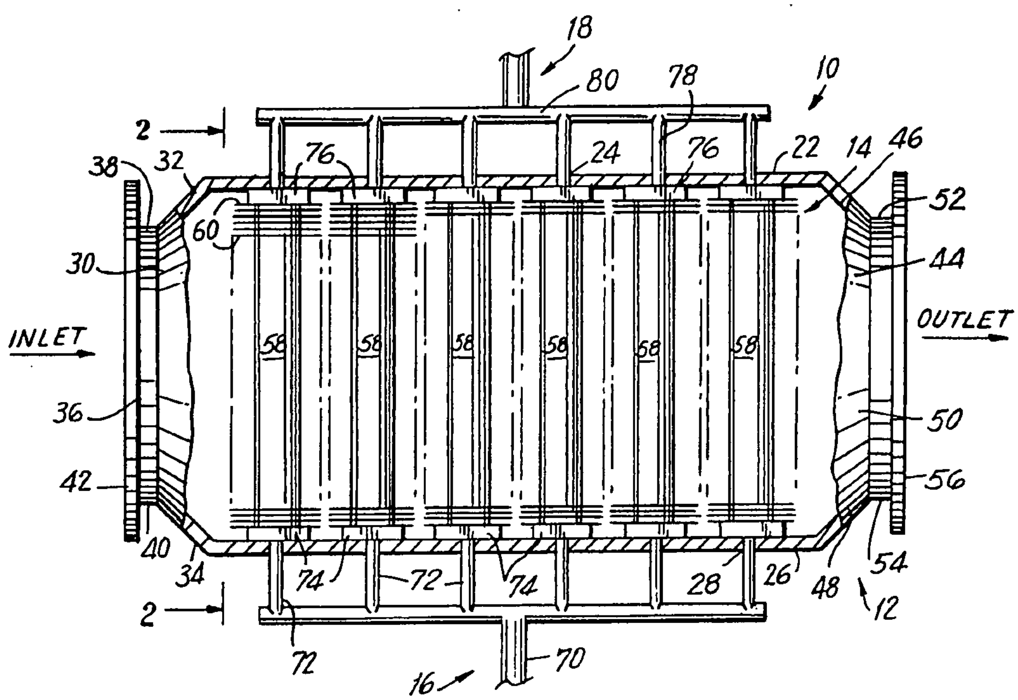
FIG. 1



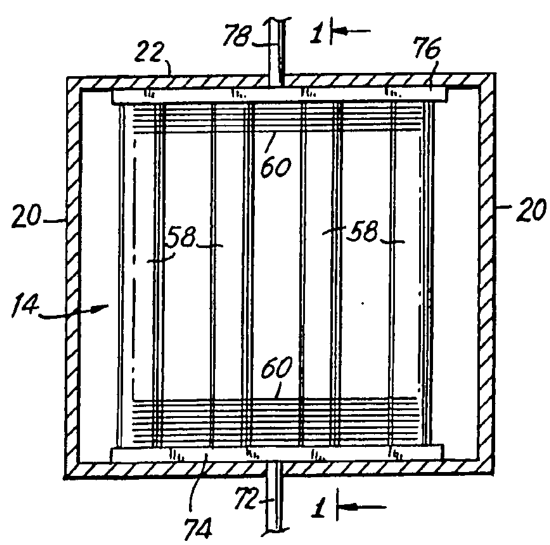


FIG. 2

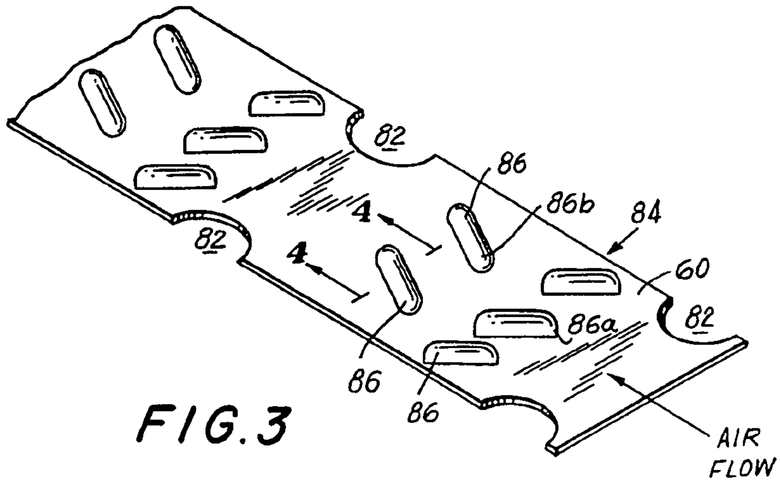


FIG.3

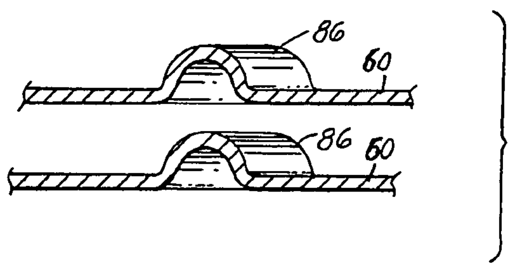


FIG.4

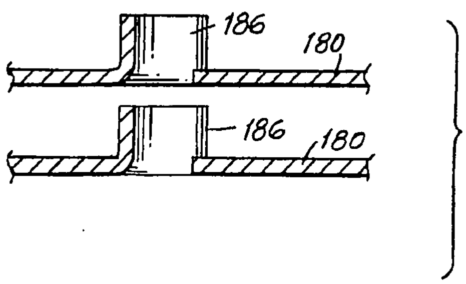


FIG.5

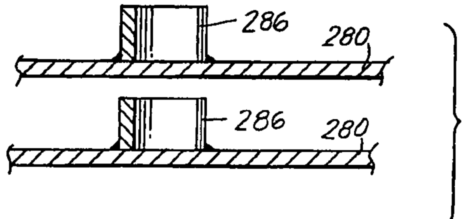


FIG.6

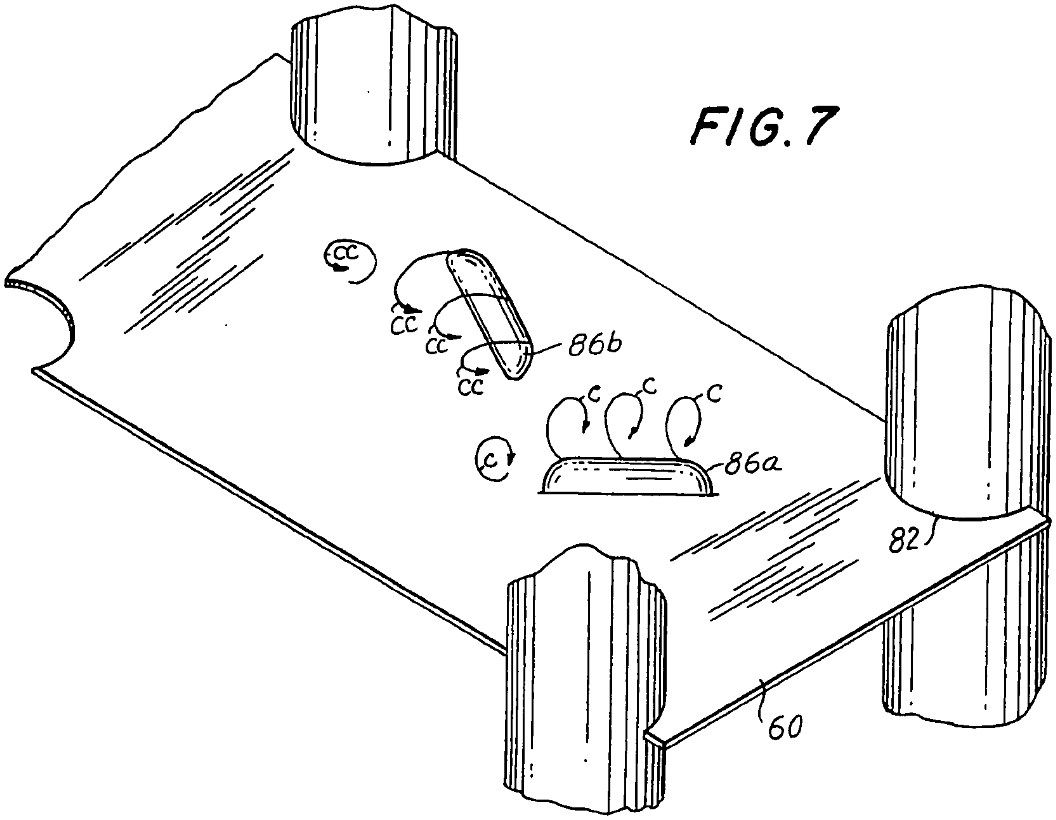
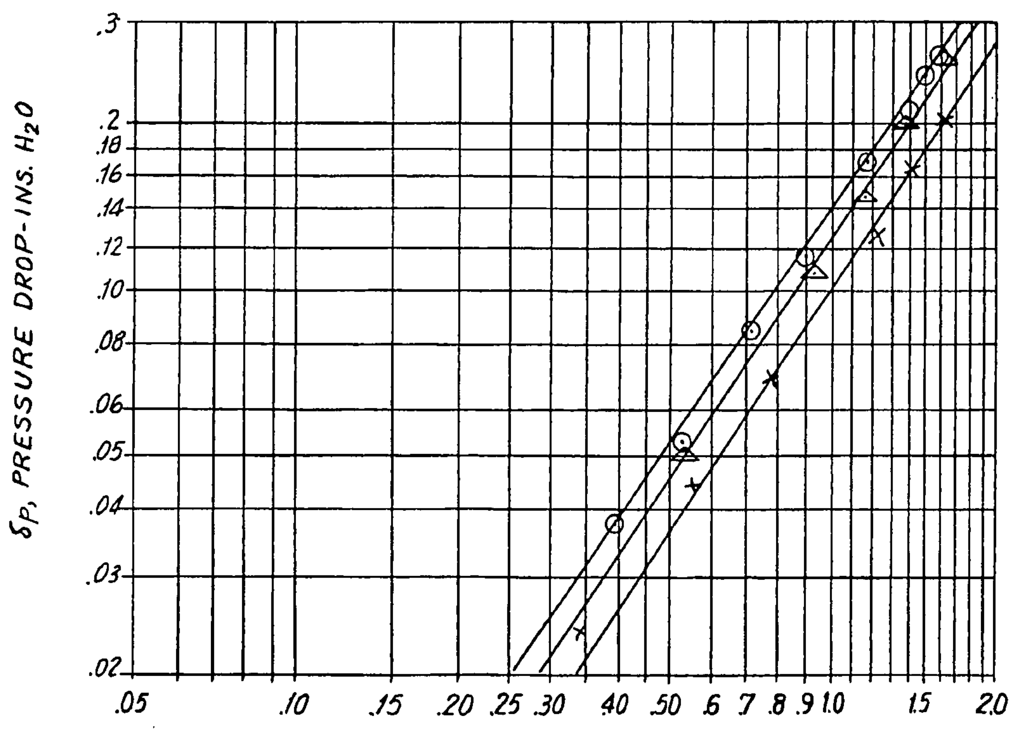


FIG.7

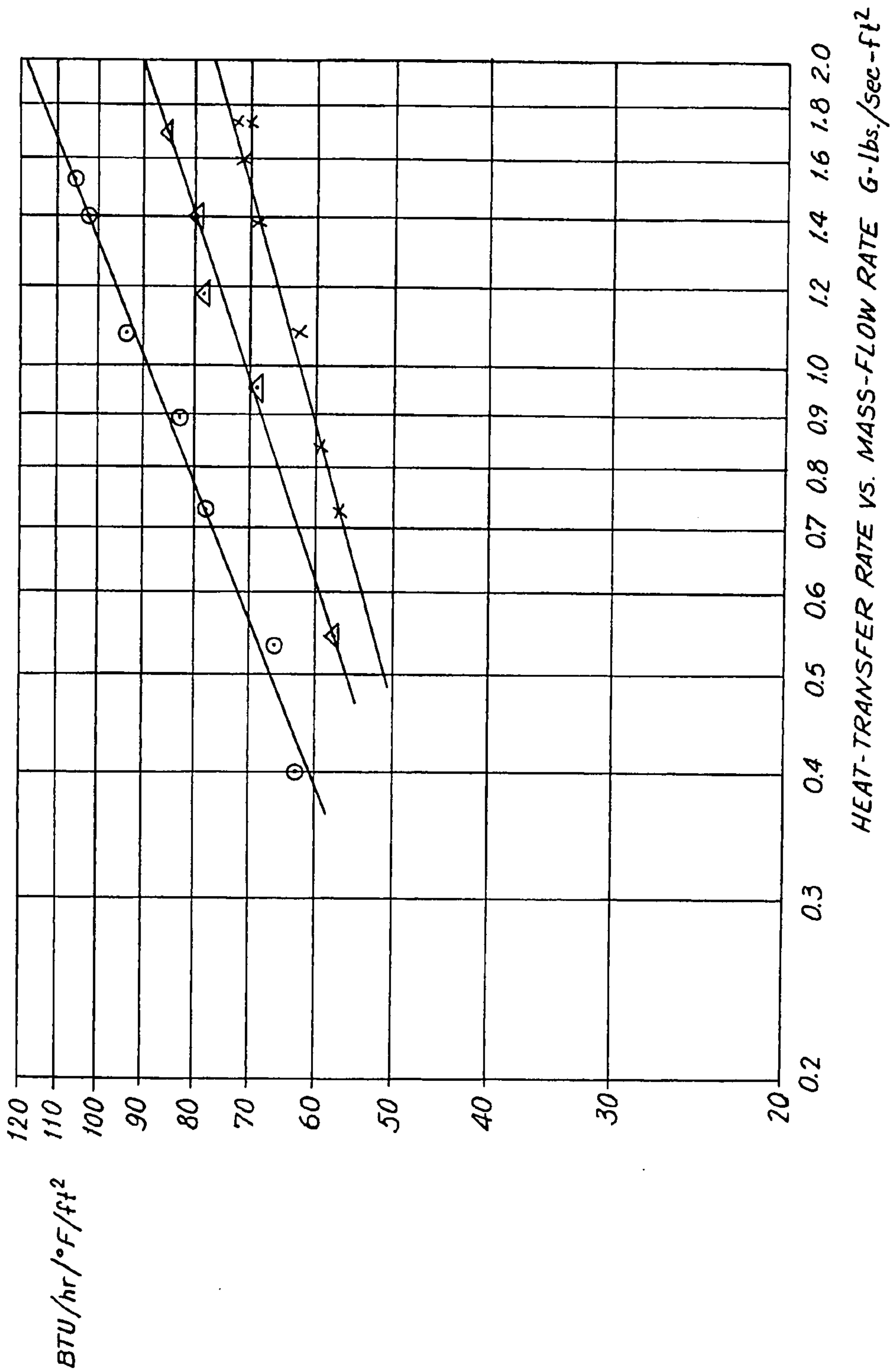
⊙ 2 SETS V.G.'S.
▽ UP STREAM V.G.'S.
X PLAIN FIN TUBES

FIG.9



PRESSURE-DROP VS. MASS-FLOW RATE G-lb/sec.-ft^2

FIG. 8


⊙ 2 SETS V.G.'S.
▽ UP STREAM V.G.'S.
× PLAIN FIN TUBES
BTU/hr/°F/ft²
120
110
100
90
80
70
60
50
40
30
20
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
1.2
1.4
1.6
1.8
2.0
HEAT-TRANSFER RATE VS. MASS-FLOW RATE G-lbs./sec-ft²

SPECIFICATION

Plate fin tube assembly and heat exchanger assembly employing same

This invention relates to heat transfer, and more particularly to a plate fin tube for assemblage as part of a heat exchanger assembly, as well as a heat exchanger assembly employing same.

In the last decade, great emphasis has been placed upon energy conservation, and in particular to apparatus and processes for reducing energy requirements whether in the form of energy saving or more efficacious use of energy. Heat exchange apparatus, particularly of the shell and tube type exchangers or air-cooled exchangers have employed finned tubes, generally of the helical fin tube type (e.g. wrapped fin "L" or applied fin "G") to obtain improved heat transfer relationships, i.e. by increasing the heat transfer surface, although single strips of metal have been fabricated to form substantially flat heat transfer fins arranged in uniformly spaced relationship and in parallelism one to the other, such as used in space heaters for home use.

In U.S. Patent Specification No. 1,966,906, there is disclosed a refrigeration coil having a series of spaced apart plate fins each of which is corrugated transversely to the normal flow or air to create a serpentine path of air flow.

Additionally, baffle devices have been employed in heat exchanger assemblies, particularly of the shell and tube type, to effect tubulent flow of a fluid or fluids being passed through such a heat exchange assembly thereby to increase heat transfer rates, but with concomitant substantial increases in pressure drops. Heat exchange assemblies employing modified baffle device have been disclosed which compensate for such increases in pressure drops (see U.S. Patent Specifications Nos. 3,708,142, 4,127,165 and 4,136,736). Generally, the necessity to generate or cause tubulent fluid flow within heat exchanger assemblies is an accepted design consideration.

The present invention aims to provide a plate fin having fluid flow interference or vortex generator means disposed thereon for use in the manufacture of a plate fin tube assembly in heat exchanger assemblies.

According to one aspect of the invention, there is provided a plate for a plate fin tube assembly, which comprises a plate member formed with at least one tube receiving opening and having a plurality of surface members extending from said plate member.

Another aspect of the invention provides a plate fin tube which comprises a plurality of spaced apart plate members connected for conductive heat transfer to a tube, each said plate member being formed with a plurality of surface members extending from said plate member.

A further aspect of the invention provides a plate fin tube assembly consisting of a plurality of plate fin tubes as above described in which the plate members are disposed in columns and rows, the surface members being disposed between the tube columns.

Yet another aspect of the invention provides a heat exchanger assembly having inlet and outlet conduits for a first heat transfer fluid and inlet and outlet conduits for a second heat transfer fluid, and further comprising a plate fin tube assembly including a plurality of plate fin tubes, each of said plate fin tubes consisting of a plurality of spaced apart plate members formed with a plurality of surface members extending from said plate members, said plate members being disposed in planar relationship and wherein said tubes are disposed in columns and rows, said surface members being disposed between tube columns.

The invention will now be further described, by way of example, with reference to the drawings, in which:-

Figure 1 is a side sectional elevational view of a heat exchanger assembly employing plate fin tube assemblies according to the present invention taken along the line 1-1 in Figure 2;

Figure 2 is a front sectional view of the heat exchanger assembly of Figure 1 taken along the line 2-2 thereof;

Figure 3 is an isometric view of a plate fin illustrating one embodiment of the fluid flow interference or vortex generator means according to the present invention;

Figure 4 is a partial sectional view of Figure 3 taken along the line 4-4 thereof;

Figure 5 is a partial sectional view of another embodiment of a fluid flow interference or vortex generator means according to the present invention;

Figure 6 is a partial sectional view of still another embodiment of a fluid flow interference or vortex generator means according to the present invention;

Figure 7 is an enlarged isometric view visualizing the vortices generated by the fluid flow interference or vortex generator means according to the present invention;

Figure 8 graphically illustrates the heat transfer rate as a function of mass flow of a heat exchanger assembly made in accordance with the present invention when compared with a heat exchanger assembly having plain fins; and

Figure 9 graphically illustrates pressure drop versus the rate of mass flow employing the same prior art heat exchanger assembly.

Referring now to Figures 1 and 2, there is illustrated a heat exchanger assembly generally indicated at 10 consisting of a shell assembly 12, a tube assembly 14, an inlet manifold assembly 16 and an outlet manifold assembly 18.

The shell assembly 12 is generally rectangular in shape and is formed of side walls 20, a top wall 22 including conduit orifices 24 and a bottom wall 26 including conduit orifices 28. At the inlet end of the shell assembly 12 and extending inwardly from the side walls 20, the top wall 22 and the bottom wall 26, there are mounted, such as by welding, intermediate side walls 30, a top wall 32, and a bottom wall 34, respectively, mounted to end side walls 36, a top wall 38, and a bottom wall 40 forming an inlet conduit for the introduction of a fluid into the heat exchanger assembly 10. An inlet flange 42 is mounted, such as by welding, to the end walls 36, 38

and 40.

At the outlet end of the shell assembly 12, and extending inwardly from the side walls 20, the top wall 22 and the bottom wall 26, there are mounted, such as by welding, intermediate side walls 44, a top wall 46, and a bottom wall 48, respectively, mounted to end side walls 50, a top wall 52, and a bottom wall 54 forming an outlet conduit for the withdrawal of the fluid after passage through the heat exchanger assembly 10. An inlet flange 56 is mounted, such as by welding, to the end walls 50, 52 and 54.

The tube assembly 14 comprises a plurality of vertically disposed tubes 58 connected for conductive heat transfer to a plurality of horizontally disposed plate fins 60 in an array of columns and rows in a manner known to those skilled in the art. For the purpose of the present description, rows of tubes refer to the tubes aligned in planes substantially perpendicular to the flow of heat transfer fluid whereas columns of tubes refer to the tubes disposed in planes substantially parallel to such fluid flow. It will be understood that the disposition of and configuration of the tube assembly 14 in the heat exchanger assembly 10 may take any form, e.g. circular, hexagonal, etc. as well as the corresponding configuration of the receiving shell assembly 12. Additionally, each plate fin 60 may comprise subsections depending on size and other manufacturing or processing considerations, as more fully described hereinafter.

The inlet manifold assembly 16 comprises an inlet conduit 70 connected in fluid flow communication to a plurality of conduits 72 which are connected to inlet manifolds 74 associated with the inlet end of each row of tubes 58. The outlet end of each row of tubes is connected to outlet manifolds 76 in fluid flow communication with a plurality of outlet conduits 78 connected to an outlet conduit 80 thereby forming the outlet manifold assembly 18.

A portion of a plate fin 60 is illustrated in Figure 3, and is formed with cut-out portions 82 for positioning of heat transfer tubes 58 and with a plurality or set of fluid flow interference or vortex generator means, generally indicated as 84, composed of, e.g., stamped or embossed, or raised, surface members 86. The sets 84 of the raised surface members 86 are illustrated as being positioned between the columns of the tubes, and each of such members is of a design configuration of an angle to the local flow direction of flow of the fluid, generally of from 10° to 35°, preferably 20° to 25°, as more fully hereinafter discussed.

Figure 4 is a cross-sectional view of an embossed or raised surface member 86 of Figure 3. In Figure 5, there is illustrated in section another type of fluid flow interference or vortex generator means 186 of the present invention formed by die stamping a portion from the plates 160 and upturning such portion on the plates 160. In the embodiment of Figure 5, it is noted that the upturned portion is positioned downstream of the portion cut-out from the plate fin. In Figure 6, there is illustrated a fluid flow interference or vortex generator means 286 formed by mounting, such as by welding, elongated plates to the plate fin 280.

Figure 7 is an enlarged visual representation of the vortices generated by the fluid flow interference or vortex generator means on one plate associated with a tube. It will be noted that the generated vortices have an axis generally parallel to the flow of fluid initially entering the chamber in which are disposed plate fin tube assemblies of the present invention. The flow of the vortices is generally of spiral configuration. As the vortices approach the tubes, they are directed away from the tubes and inwardly between the tubes, and thus in this context reference is made to the local flow direction, particularly considering multiple tube columns and multiple vortex generator means.

While the plate fins 60 are connected to the tubes 58, as hereinabove described, the following description relates to a plate fin-tube assembly having plate fins and tube configurations of the present invention contrasted to fin tubes of the hereinabove mentioned prior art. Generally, the plate fins to be connected to the tubes in conductive heat transfer relationship are rectangularly shaped and affixed to tubes preferably of a cross-sectional shape other than circular, e.g. oval, elliptical, or rectangular, preferably elliptical. The minimum distance between the vortex generator means establishes the minimum distance between the fins. The height of a vortex generator should be approximately one-half the distance between the adjacent plate fins, with the distance between fins preferably being not greater than the distance between the generated vortices.

It is preferable to locate the vortex generator means behind the leading edge of the plate fin so as not to interfere with fluid flow at that point. Additionally, the generated vortices should not interfere with the collar vortices developed about the tube. Preferably the vortex generator means are positioned on the plate fin to form juxtaposed vortices of counter-rotating flow directions. Note in Figure 7 wherein the vortex generated by the vortex generator 86*a* is in a clockwise fluid flow direction (illustrated by the arrows "C") whereas the vortex generated by the vortex 86b is in a counterclockwise fluid flow direction (illustrated by the arrows "CC"). Such a positioning of vortex generator means to generate vortices of opposite flow direction next to one another has an overall boosting effect to provide for better heat transfer rates at minimal reductions in pressure drop. Adjacent vortices of like fluid flow direction, particularly of close proximate distance, should be avoided since the net effect is instability as a result of coalescences of such vortices. Furthermore, the spacings of the vortex generator means should not be too close together, even with clockwise-counterclockwise relationship, since the effectiveness of the vortices are reduced as a result of inter-destruction. It will be understood that, at too low fluid flow rates, the efficacy of vortex generator means would be questionable unless such heat transfer assembly were to contemplate other duties involving efficacious use of vortex generator means.

To more fully understand the present invention, the following example is provided.

Example

Aluminum plate fins, 1.016 inch in thickness and spaced at eight to the inch soldered to rectangular tubes. The plate fins are provided with a fluid flow interference means of the elongated embossed type (0.3" long and 0.0625" in height) of Figures 3 and 4. The plate fin and tube assembly is positioned within a shell assembly. Steam is passed through the tubes and air drawn through the plate fin and tube assembly at varying air mass flow rates. By measuring the rise in air temperatures at given rates of mass flow (G = lb./ft.^{2}/sec.), it is possible to calculate the rate of heat transfer (U-BTU/hr. F°/ft.^{2}) for various mass flow rates.

Figures 8 and 9 graphically illustrate the results of the tests in which the overall heat transfer coefficient U; pressure drop ςp were determined as functions of the shell side flow rate G, the appropriate methods of calculation known to one skilled in the art from data taken during comparable test runs.

The heat exchanger, designated by the circle in the graphs, is provided with the sets of fluid flow interference or vortex generator means of Figure 3. A heat exchanger, designated by the triangle in the graphs, is provided with a set of upstream fluid flow interference or vortex generator means only, whereas a heat exchanger with plain fins is designated in the graphs by an "X".

From Figure 8, it is readily seen that the heat transfer rate is appreciably increased by use of the fluid flow interference or vortex generator means of the present invention. At typical mass flow rates of 1.2 lb/ft.^{2}, the inrease is about 50%. It is noted that the difference in the slope on the log/log plot, i.e., difference indices for G, suggests significant changes in the type of air flow as a result of the fluid flow interference or vortex generator means of the present invention.

A reduction in the improvement (in heat transfer) due to the vortex generator means as Reynolds number falls is expected from the reduction in the persistance of the vortices that will accompany the increase in the relative importance of viscosity as the Reynolds number becomes smaller. For any given number of sets of fluid flow interference or vortex generator means, there will be an optimum position for the sets from a heat transfer point of view, and by proper balance between heat transfer rates and pressure drops, it is possible to optimize the number of sets of vortex generator means as well as the configuration for each set.

From Figure 9, it can be seen that the use of vortex generator means results in an increase in pressure drop and hence in cooling power. It is to be noted that ςp for the plain plate fins is measured cold instead of hot as in all other cases, which could mean that the pressure drop for such conditions might be about 5% more than that shown. What is clear, however, is that, for a value of G of at least 1.2, the increase in power due to the use of vortex generator means is a good deal less than the increase in heat transfer.

While the plate fin tube of the present invention has been discussed with reference to assemblage of a plurality thereof for incorporation into a shell to form a shell and plate fin tube type heat exchanger, it will be understood that such plate fin tubes may be used in other forms of heat exchanger assemblies, e.g. air-cooled heat exchangers of the "A" frame type with fan, or natural draft dry cooling towers and the like. Thus, at given performance requirements, it is possible to construct a heat exchanger assembly employing the plate fin tubes of the present invention having higher heat transfer rates when compared to a heat exchanger assembly having plain fin tubes.

It will be apparent to one skilled in the art that the present invention has the effect of causing the heat transfer fluid to becomes more turbulent, but that such turbulence is of a uniform nature.

While the invention has been described in connection with an exemplary embodiment thereof, it will be understood that many modifications will be apparent to those of ordinary skill in the art and that this application is intended to cover any adaptations or variations thereof. Therefore, it is manifestly intended that this invention be only limited by the claims appended hereto.

CLAIMS

1. A plate for a plate fin tube assembly, which comprises a plate member formed with at least one tube receiving opening and having a plurality of surface members extending from said plate member.

2. A plate for a plate fin tube assembly according to claim 1, wherein said surface members are formed by embossing said plate member.

3. A plate for a plate fin tube assembly according to claim 1, wherein said surface members are formed by upturning sections of said plate member.

4. A plate for a plate fin tube assembly according to claim 1, wherein said surface members comprise strip members mounted on said plate members.

5. A plate for a plate fin tube assembly according to any preceding claim, wherein said surface members are at an angle of from 10° to 35° to the intended local flow direction.

6. A plate for a plate fin tube assembly according to claim 5, wherein said angle is from 20° to 25°.

7. A plate for a plate fin tube assembly according to any preceding claim, wherein said tube receiving opening is elliptically-shaped.

8. A plate for a plate fin tube assembly according to any preceding claim, wherein said surface members are disposed in uniform array.

9. A plate for a plate fin tube assembly according to any preceding claim, wherein said surface members are disposed to form adjacent generated vortices of opposite rotation.

10. A plate fin tube which comprises a plurality of spaced apart plate members connected for conductive heat transfer to a tube, each said plate member being formed with a plurality of surface members extending from said plate member.

11. A plate fin tube according to claim 10, wherein said surface members are at an angle of from 10° to 35° to the intended local flow direction.

12. A plate fin tube according to claim 11, wherein said angle is from 20° to 25°.

13. A plate fin tube according to any one of claims 10 to 12, wherein said tube is elliptically-shaped.

14. A plate fin tube according to any one of claims 10 to 13, wherein said surface members are disposed in uniform array.

15. A plate fin tube according to any one of claims 10 to 14, wherein said surface members are disposed to form adjacent generated vortices of opposite rotation.

16. A plate fin tube according to any one of claims 10 to 15, wherein said surface members are disposed such that axes of generated vortices are directed around said tube.

17. A plate fin tube according to any one of claims 10 to 16, wherein the height of said surface members from said plate members is about one-half the distance between adjacent plate members.

18. A plate fin tube assembly for a heat exchanger assembly consisting of a plurality of plate fin tubes as claimed in any one of claims 10 to 17, wherein said plate members are disposed in planar relationship and wherein said tubes are disposed in columns and rows, said surface members being disposed between tube columns.

19. A plate fin tube assembly according to claim 18, wherein said surface members are disposed behind the leading edge of said plate members.

20. A plate fin assembly according to claim 18 or claim 19, wherein said surface members are disposed such that the rotation of generated vortices is directly around said tube.

21. A heat exchanger assembly having inlet and outlet conduits for a first heat transfer fluid and inlet and outlet conduits for a second heat transfer fluid, and further comprising a plate fin tube assembly including a plurality of plate fin tubes, each of said plate fin tubes consisting of a plurality of spaced apart plate members formed with a plurality of surface members extending from said plate members, said plate members being disposed in planar relationship and wherein said tubes are disposed in columns and rows, said surface members being disposed between tube columns.

22. A heat exchanger assembly according to claim 21, wherein said surface members are at an angle of from 10° to 35° to the intended local flow direction.

23. A heat exchanger assembly according to claim 22, wherein said angle is from 20° to 25°.

24. A heat exchanger assembly according to any one of claims 21 to 23, wherein said surface members are disposed in uniform array.

25. A heat exchanger assembly according to any one of claims 21 to 24, wherein said surface members are disposed to form adjacent generated vortices of opposite rotation.

26. A heat exchanger assembly according to any one of claims 21 to 25, wherein said surface members are disposed such that axes of generated vortices are directed around said tubes.

27. A heat exchanger assembly according to any one of claims 21 to 26, wherein said surface members are disposed behind the leading edge of said plate members.

28. A heat exchanger assembly according to any one of claims 21 to 27, wherein said plate members are substantially parallelly-disposed to one another and said tubes extend substantially perpendicular to said plate members and in conductive heat transfer relationship thereto.

29. A heat exchanger assembly according to any one of claims 21 to 28, wherein said tubes are at least substantially rectangular.

30. A plate for a plate fin tube assembly substantially as described herein with reference to the drawings.

31. A plate fin tube substantially as described herein with reference to the drawings.

32. A plate fin tube assembly substantially as described herein with reference to the drawings.

33. A heat exchanger assembly substantially as described herein with reference to the drawings.

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